RECIPROCATING EXPANSION ENGINES WITH UNLUBRICATED CYLINDERS

Prepared by W. A. Morain
The Cooper-Bessemer Corp.
Mount Vernon, Ohio

Presented by T. White
The Cooper-Bessemer Corp.
Mount Vernon, Ohio

There is a steadily growing interest in unlubricated cylinders for chemical process compressors and expanders. The interest is particularly keen in expansion engines for cryogenic service. The complete removal of entrained oil from gas streams has never been an easy task, and at cryogenic temperatures the problem is even tougher. For continuous operation, duplicate filters and adsorbers must be provided, with means for switching back and forth to clean and reactivate the elements. When this equipment is located in very cold gas streams, it is almost impossible to avoid serious heat leaks and a high degree of inaccessibility. This, in turn, results in inefficiency and higher maintenance costs.

Unremoved oil

If the oil is not completely removed, the resultant contamination of the downstream product and equipment is only one of the serious consequences. Worse still is the explosive hazard which exists wherever oxygen concentrations are encountered, for example, in the almost countless air separation plants in service all over the country.

Reciprocating expansion engines with unlubricated cylinders are not new, and many such units have given satisfactory service, for example, those employed by the Bureau of Mines at helium extraction plants, and others used in hydrogen and helium liquefaction service. Most of these units, however, operate under limited service conditions where inlet pressures and piston speeds are severely curtailed. The severe duty requirements of the larger air separation and liquefaction plants have recently been tackled in earnest, and this is the field with which we are principally concerned. Recent breakthroughs in design techniques and materials are putting the unlubricated cylinder in a position of ever widening acceptance.

Design features

The designer of this type of equipment should be prepared to handle the following problems:

a) Means for preventing the entry of any lubricating oil into the nonlube area from adjacent compartments.

b) Means for preventing the corrosion of all parts in the nonlube area and connecting passages.

c) Means for ensuring suitable service life of the nonlube working parts and mating surfaces.

Exclusion of oil

The most probable source of oil entry into the nonlube area is along the piston rod. If any one element of the rod can travel alternately in the lubricated and nonlube areas, some oil transfer is bound to occur. The answer in this case, and quite old in the art, is to furnish a distance piece to go between the crosshead housing and the cylinder. This, in turn, requires an elongated rod and an extra element of rod packing.

In some cases, a slinger ring is attached at an intermediate location on the rod to throw off any oil which may have penetrated to this point. Figure 1 shows a section through a conventional, lubricated cylinder, and Figure 2 shows the modifications for the nonlube design. The rod packing at the cylinder end is not lubricated, and may use filled Teflons or other nonlube materials. Typical rod packings are shown in Figure 3.

Corrosion

It is surprising how fast everything in the cylinder starts to rust once the protective coating of lubricating oil has been removed. This is particularly noticeable in expansion engines, since conventional drying agents do not produce dew points in the working fluid low enough to prevent some condensation at cryogenic temperatures. The best answer to this problem is to use noncorrosive materials, or to give all corrodible parts a protective coating. This treatment should also extend to gas passages and associated piping. Wherever atmospheric vents are used, they should be provided with nonreturn check valves to exclude the possibility of moisture entry. Failure to eliminate all corrosion can result in accelerated wear and product contamination.

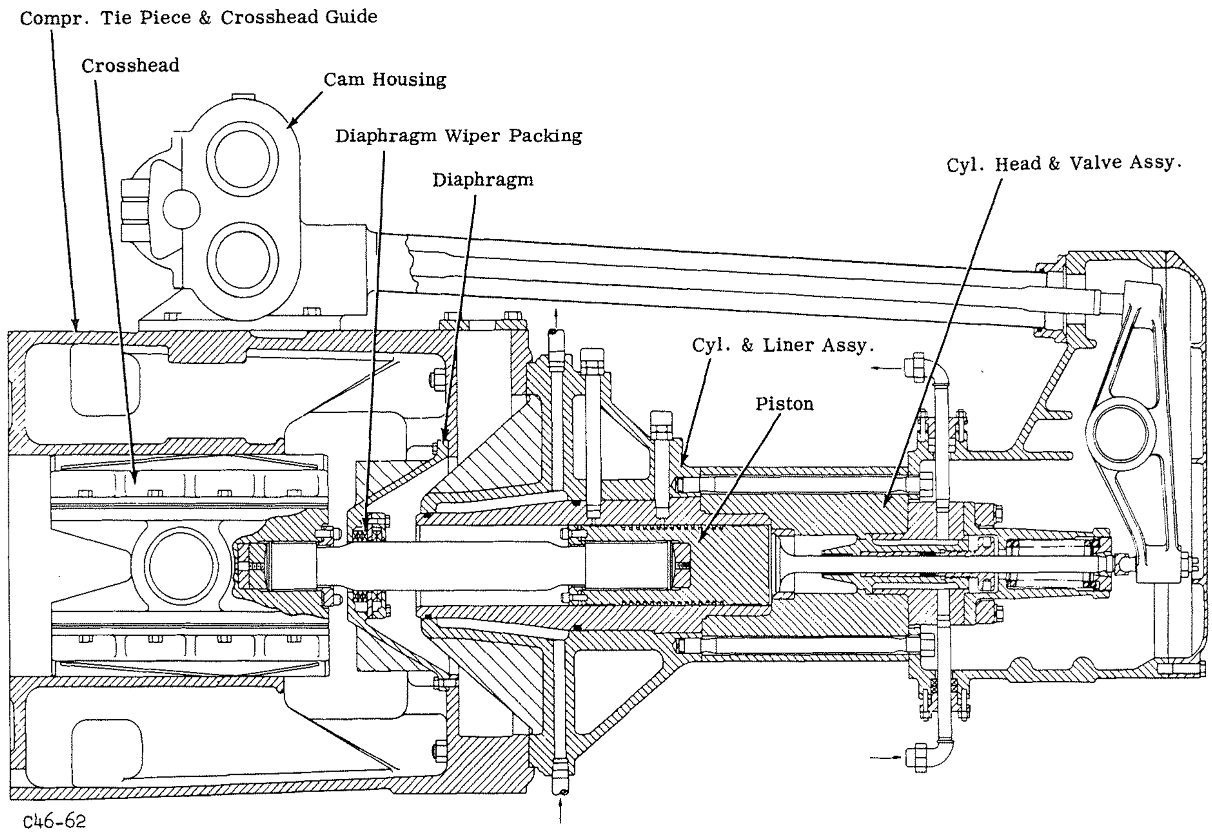

Figure 1. Cross section of lubricated expansion engine.

Wear of working parts

The service life of the working parts in unlubricated cylinders has not yet equalled that experienced in lubricated cylinders under similar operating conditions.

It is most difficult to predict the probable life of any combination because of the substantial effects of a number of factors over which the designer may have little or no control. Once suitable designs and materials have been selected, it is imperative that proper attention be given the field assembly and operation if success is to be assured.

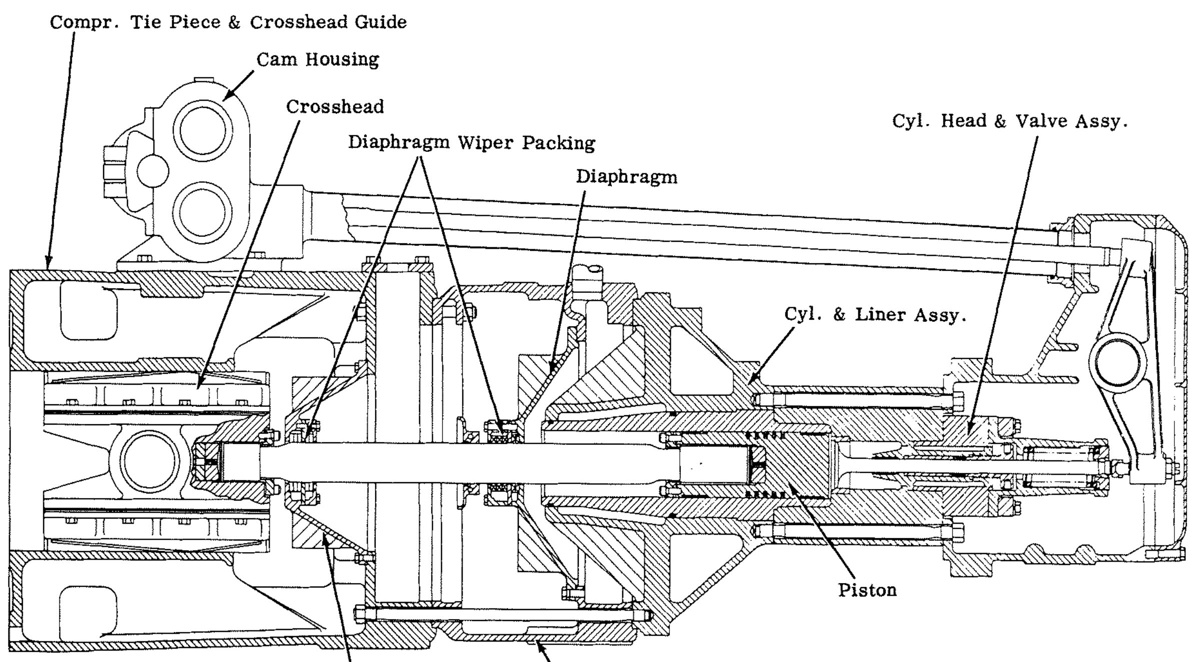

Figure 2. Cross section of nonlubricated expansion engine.

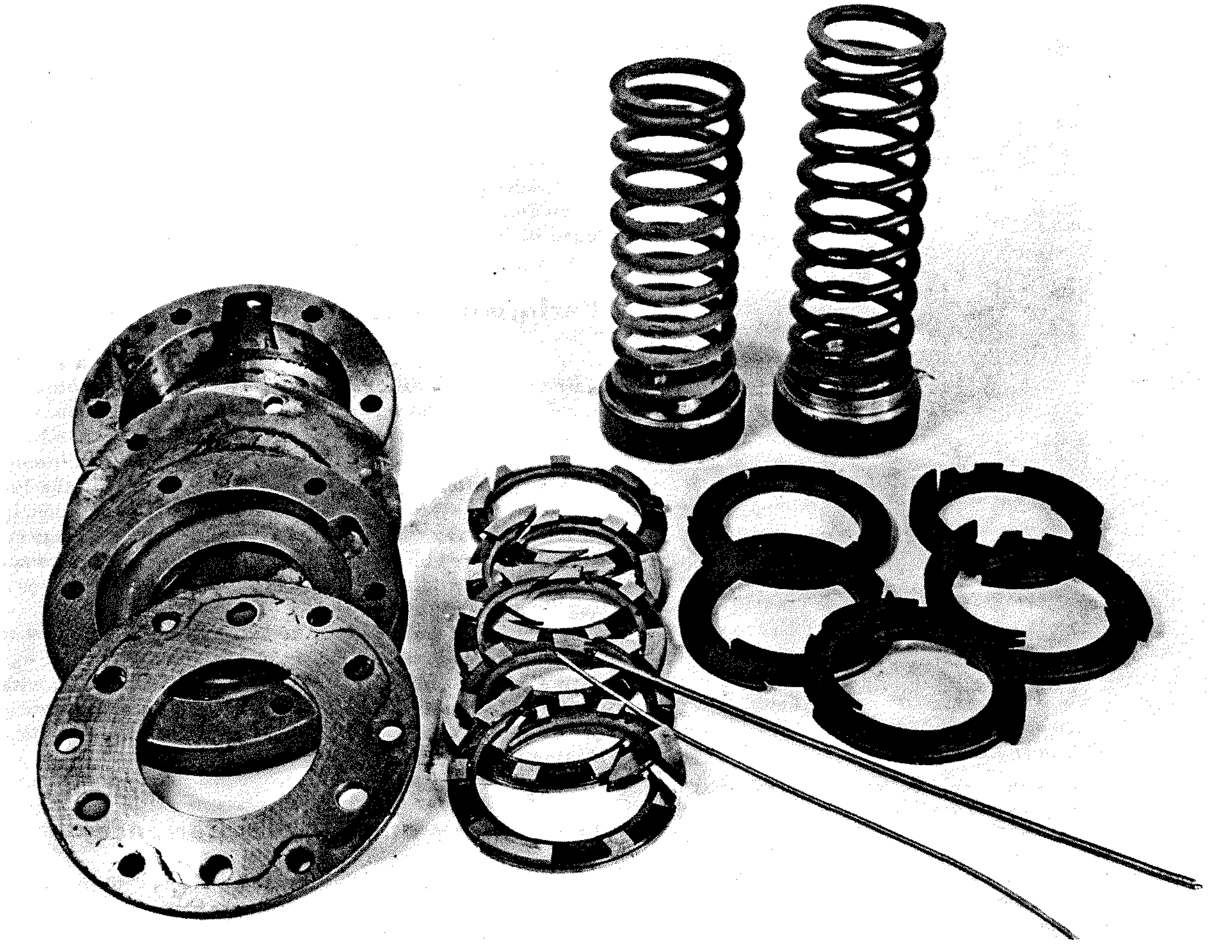

Figure 3. Typical rod packings.

Pressure loading

It is obvious that there are limits to the pressures and rubbing speeds which can be handled in nonlube cylinders. However, it does not appear that any satisfactory criteria have thus far been established. The often used (and misused) PV factor, based on the mean cylinder pressure and the mean piston speed is virtually meaningless. It might be of some use if we had but one piston ring and a cylinder without blowby. As soon as one introduces a plurality of piston rings, and considers any leakage, one no longer knows what pressure loading any one ring supports. Suppose the PV factor for a given cylinder is 300,000 (500 lb./sq. in. mean pressure and 600 ft./min. piston speed). If one has 6 rings in complete pressure balance, the individual PV factor would be only 50,000.

Relatively complete pressure balance has been achieved by recent design techniques. In addition to this, there are other techniques involving bull rings and labyrinths ahead of the conformable rings. All of these schemes have merit, and they serve to extend the permissible cylinder loading. It is felt that the simple and conventional ring is out of place in highly loaded nonlube cylinders.

Wear bands are in wide use in nonlube cylinders. We prefer the continuous "rubber band" type of ring, in order to avoid pressure loading. Even with this type, simple pressure relief notches at the leading and trailing edges of the rings are desirable. The static loading supported by wear bands is usually limited to 10 to 15 lb./sq.in., based on the developed area of a 120° arc of circumference.

A nonlube piston, as it was removed from an engine after a test run, is shown on Figure 4. The method of assembly used to install the wear bands is shown in Figure 5.

Piston ring materials

The materials used for piston rings in unlubricated cylinders are quite varied. At lower pressures, carbon, laminated resins, and even leather have been found to give satisfactory service. At higher loadings one finds filled Teflons widely used. Filling agents such as carbon, graphite, glass fiber, molybdenum disulfide, and bronze each have particular advantages in specific services. Some success has been experienced with Kel-F impregnated with Teflon and other fillers. At present, the bronze-filled Teflons are considered to be in the forefront for general application.

The cylinder liner finish and hardness are also important, and it is our feeling it cannot be too hard or too smooth. There are others who prefer a given microfinish in the liner wall, say from 10 to 20, in order to provide a matrix for the Teflon which the wall is supposed to retain.

For valve stem packings, and wherever else a readily conformable packing is required, braided Teflon is equal to the occasion. The fact that it requires neither heating nor lubrication makes it all the

Figure 4. A nonlube piston removed from an engine after a test run. FXA piston, bottom view, 175 hr.

more attractive. However, at very cold temperatures, we do recommend some heat to simplify the follow-up tightening on the gland.

Foreign matter

The use of a high grade inlet liner filter, with filtration, in depth, down to 2 to 5 microns, is a most significant factor in promoting ring life. The nonlube cylinder has no oil film to cushion and wash away the dirt. Particles of foreign matter which may be plastic or even oily at room temperature may be downright abrasive at cryogenic temperatures. In most of the cases where nonlube cylinders have failed to operate as expected, one has been able to trace the difficulty to foreign matter in the gas. Microscopic examination of the ring surface and cylinder deposits after premature failure may reveal silica, alumina, and charcoal, to say nothing of pipe scale, and welding beads.

Operating practices

Since Teflon has poor dimensional stability, it should not be allowed to overheat. It is recommended, therefore, that nonlube cylinders be equipped with a temperature indicating device in the liner wall. This should be continuously monitored, and provided with an alarm or shutdown device in case of excess wall temperature. When wearing in new rings or making drastic changes in operating conditions, the wall temperature will often give the first indication of erratic ring performance. Similarly, when the rings approach the end of their useful life and begin to blow by, a marked drop in wall temperature should be observed.

A new ring setup should be given a brief wearing-in period, certainly at reduced pressures, and if possible at reduced speeds. At ultra-low temperatures, it may take quite a while to achieve good conformability with the liner walls. In this case, valuable time can be saved by making temporary arrangements to do the wearing-in at some higher temperature.

The effectiveness of even the best gas filters may be substantially reduced if they are not properly maintained. A dependable, differential-pressure gauge should be connected across inlet and outlet and the manufacturer's recommendation for permissible pressure difference should never be exceeded.

Performance comments

The nonlube cylinder will usually show a small gain in thermal efficiency and a small drop in volumetric. Since there is no need to heat the cylinder walls with warming water (as is customary with lubricated cylinders), this source of heat leak is eliminated. The dry piston ring cannot seal quite as well as the lubricated one. Moreover, to avoid excessive friction, it is customary to use fewer rings in nonlube designs. However, any volumetric deficiency can easily be made up by slightly delaying the "cutoff" or inlet valve closing point.

The cleanliness of the exhaust gas from the nonlube cylinder is one of its outstanding characteristics. The elimination of the downstream filters, with their installation and maintenance problems, easily justifies the moderate increase in cost of the nonlube design.

Laboratory and field experience

The material presented here is based on intermittent laboratory experiences covering a 4-year period, and 2 1/2 years in the field, working at inlet

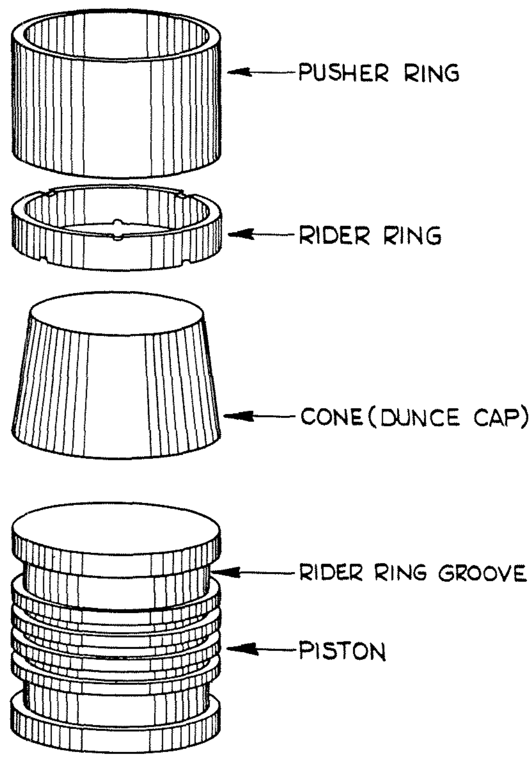

Figure 5. The method of assembly used to install wear bands.

pressure levels from 2,000 to 3,000 lb./sq. in. and piston speeds from 450 to 600 ft./min. Most of the experience to date has been with air, nitrogen, helium, and hydrogen as the working fluids. We have not seen any special problems directly attributed to the gases themselves, except of course the sealing problems introduced when using the lighter gases. Since hydrogen and helium service usually involves ultra-low temperatures, the increased hardness of Teflon in this range should permit higher unit pressures for more effective sealing.

Reciprocating expansion engines with unlubricated cylinders are out of the laboratory, and available for service wherever a clean oil-free downstream product is required. While the ring life still lags behind that for lubricated service, breakthroughs are continually being experienced in designs and materials, and the future is promising. The present status is aptly described by a customer who states that the reciprocating expansion engine with unlubricated cylinder may definitely be regarded as an acceptable piece of process equipment.

DISCUSSION

WALTON—Sun Olin: Mr. White, how much blow-by in terms of percentage of gas flow (or quantity) expanded from 400 lb. to 200 lb. would you feel is a reasonable amount?

WHITE: First off, 400 lb./sq. in. is quite low. I can give you the actual percentage of blow-by when the inlet pressure is say 2,500 or 3,000 lb./sq. in. Ordinarily, it runs about 0.2% to about 0.8% while expanding air. This is with new rings which have had an opportunity to seat in. What the blow-by would be after the rings had run for several thousand hours might be quite another matter. Another element that enters into the amount of blow-by, is the design of the ring itself. We can purposely put blow-by into the ring, and this is done in order to cool the cylinder liner wall temperature—with the hope of keeping the piston rings as cool as possible, within certain limits of course. We believe that you have longer ring life if the rings are kept cool.

WALTON: Are you speaking of single-acting cylinder blow-by?

WHITE: That is correct—coming in at say around 2,500 to 3,000 lb./sq. in. and discharging at about 85 to 100 lb./sq. in.

WALTON: How about helium—do you feel as though this is still a reasonable percentage of blow-by for helium?

WHITE: We did run on our closed test loop last winter on helium, at 2,000 lb./sq. in. abs. at -410 °F outlet temperature. The helium blow-by that we experienced with rings which we do not consider to be the best ring design today, was on the order, as I recall, of about 0.7%. We do today, however, have a ring which we are confident will seal considerably better than that.

WALTON: How about valve packing? We have vertical nonlubricated expanders and have had problems of excessive blow-by and also excessive leakage from valve packing, which is, of course, another point where you lose refrigeration. We have tried square cut Teflon braided packing, chevron packing of Teflon, and leather packing with very little success.

The trouble is either you have great leakage and when you start to snug up on the packing, the valve hangs up. Do you have any solution to this problem?

WHITE: What gas and what temperature are you speaking of?

WALTON: Well, we're expanding hydrogen from about 440 to 200 lb./sq. in. gauge and the exit temperature is -333 °F.

WHITE: What is the inlet temperature and are you having difficulty with both the inlet and the exhaust valves?

WALTON: Yes, both valves; inlet temperature is about -295. We're getting about 40° drop in expansion.

WHITE: Speaking strictly of air separation machines operating with 2,500 to 3,000 lb./sq. in. inlet pressure, and with approximately 0 °F to -50 °F inlet temperature, our experience with inlet valve stem leakage, when using braided Teflon, has taught us to use warm water (i.e., about 140 °F) around the packing cage. The heat keeps the Teflon flexible enough to conform to the valve stem and to the cage and, thereby, effect a good seal. One of our engines ran for more than 3,400 hr. without the inlet valve stem packing being tightened.

Braided Teflon exhaust valve stem packing, on the other hand, has sealed perfectly satisfactorily without warm water or any other kind of heat being circulated around the packing cage. The pressure is usually on the order of 85 to 100 lb./sq. in. and the temperature is in the vicinity of -260 °F to -280 °F. But inasmuch as we do heat the inlet packing cage, we also circulate warm water around the exhaust cage even though it is not absolutely necessary. Though the exhaust packing seldom has to be tightened, the heat does make tightening easy. Without the heat, the Teflon gets so stiff that it does not conform readily to either the valve stem or the cage. One then runs the risk of rupturing the braid by trying to force it to conform.

On the helium engine which we ran in our lab last winter, we did encounter difficulty with the inlet valve packing. But, again, we were running at 2,000 lb./sq. in. inlet, not at 400 or thereabouts. What we would have experienced in the way of difficulty at 400 lb./sq. in., I have no way of saying. But we did have trouble at 2,000 lb./sq. in. As a consequence, we had to redesign the inlet valve packing arrangement. We now have a double-gland arrangement, rather than the customary single gland. This design also uses braided Teflon and has, likewise, not proved altogether satisfactory while being used in the field on ultra-cold hydrogen. We are now experimenting in our laboratory with several entirely different packing designs.

WALTON: One last question: What is a reasonable normal ring life, in hours, to expect?

WHITE: I presume that you are referring to completely unlubricated piston rings. Ring life is difficult to predict because there are many variables. Offhand, I would say that one of the most significant variables is geographic location. I say this on the basis of some very unhappy experiences during the past 9 months. In West Texas, where there is a lot of wind-blown sand, 3 to 5 micron in-depth filtration is mandatory. We do not consider crimped edge-type filters to be satisfactory. With 2,500 lb./sq. in. inlet pressure and in-depth filtration of the inlet air, the West Texas engine has run for as long as 3,440 hr. on one set of piston rings—and all at high cutoff.

A second variable is the type of desiccant being used upstream of the engine. If the desiccant is prone to disintegrate into a talcum-like powder, this solid contaminant must be caught with filters before reaching the engine. Again, we recommend at least 3 to 5 micron in-depth filtration. The filter elements themselves must be of a design which will not gradually spall or disintegrate as a consequence of gas line pulsation.

A third variable is piston ring surface speed and inlet line pressure. We do not have data which enables us to separate the effect of pressure and the effect of piston ring surface speed on ring wear, but we do have field data which show that relatively low surface speed and relatively low inlet pressure in combination do give longer piston ring life. One of our engines in the field runs at 394 ft./min. piston speed and with an inlet pressure which ranges from approximately 900 to about 2,000 lb./sq. in. Piston ring life on this engine has been as long as 6,250 hr.

A fourth variable is the piston ring design itself, and in this I include the kind of material as well as the shape.

WALTON: Would you say 4,000 hr. is a reasonable ring life?

WHITE: On the basis of our experiences to date, one can only conclude that no single roundhouse figure of estimated piston ring life will be valid for all installations. At the present time, with engines thought to be in reasonably acceptable condition, but not invariably equipped with the type of inlet filters which we recommend, piston ring life ranges from 1,600 hr. to 6,250 hr.

WALTON: Was there any discussion held on oil creeping along the piston rod? You described a distance piece with a deflector ring, a type of ring arrangement.

WHITE: It's what we call an oil slinger.

WALTON: How much oil have you experienced getting past that ring, going up to the cold packing area?

WHITE: I do not know. However, oil does get past the packing rings to some extent. If it didn't, you'd be running so dry you'd score the rod. You will get a microscopically thick film of oil along the rod, and the purpose of the slinger is to catch this and assure that it will not get as far as the cylinder.

WALTON: You mentioned a high temperature cylinder liner alarm previously. At what temperature is it set to go off?

WHITE: The alarm is set to go off—or shut the engine down—when the cylinder liner wall temperature reaches 220 °F. The thermocouple is located midway in the stroke of the piston, and its tip is 1/2 in. away from the wearing surface. The cylinder liner temperature is greatly affected by piston ring design and by the amount of piston ring wear. Pressure ratio, inlet pressure, the amount of cutoff, piston ring surface speed, and inlet and exhaust temperature levels will also affect cylinder liner temperature; but, in general, the cylinder liner wall temperature in air separation service usually lies between approximately 50 °F and –100 °F.